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Economic thermal insulation thickness for pipes and ducts: A review study

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ABSTRACT

Energy conservation has become an increasingly important issue for all sectors, particularly in industry. Therefore, the thermal performance of insulation systems and their influence on heat loss/gain in various applications in addition to economic considerations have received increased attention in recent years. In this study, a literature review of papers that addressed the optimum economic thickness of the thermal insulation on a pipe or duct with different geometries used in various industries was carried out. The studies related to determining the critical insulation thickness for different geometries including circular shapes were investigated. The heat transfer equations, the basic results, the optimization procedures and the economic analysis methods used in the studies were presented comparatively. Additionally, a practical application example based on optimizing the insulation thickness on a pipe was performed, and the effective parameters of the optimum thickness were investigated.

Contents

1. Introduction

Thermal insulation systems have been used in practice for many years for different applications and purposes, such as to decrease heat transfer to/from surfaces, to control the process and surface temperatures, to avoid the condensation problem, and to provide a comfortable indoor thermal environment. Increasing concerns regarding energy efficiency, climate change and awareness of the limited energy resources, the use of a proper amount of thermal insulation for buildings and industrial applications has gained popularity. When examining the sectoral distribution of the energy demand, it is seen that considerable portions of the global energy have been consumed in the residential and industrial sectors, accounting for approximately 30% and 40%, respectively, of the total global energy [1–3]. Thermal insulation is primarily used to limit heat loss/gain from/to surfaces under operating conditions at temperatures above or below ambient temperature, i.e., to provide a contribution for energy conservation. Energy conservation is a major concern in many industrial applications. The primary reasons to conserve energy include maximizing the return on investment and minimizing the life cycle cost and the emissions associated with energy consumption.

In addition to economic considerations, attention has focused on improving performance and thermal efficiency of the insulation systems in recent years. The concept of an economic thermal

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Nomenclature

Bi	Biot number ($h\ r_o/k$)
C	cost, US$/kg or US$/m^3
$\dot{C}$	cost rate, US$/s
CRF	capital recovery factor
d	discount rate
D	diameter, m
DP	ratio of down payment to initial investment
$\dot{E}$	exergy rate, kJ/s
h	convective heat transfer coefficient, W/m^{2}K
HDD	heating degree-days
Hu	lower heating value of fuel, kJ/m^3
i	inflation rate
k	thermal conductivity, W/mK
L	pipe length, m
LT	lifetime, years
M_s	ratio of first year miscellaneous costs to initial cost
N	annual operating time, hours
Pr	Prandtl Number
PWF	present worth factor
Q	heat loss rate from pipe, kJ/s
r	radius, m
R	thermal resistance, m^{2}K/W
R_v	ratio of the resale value to initial investment
Ra	Rayleigh number
Re	Reynolds Number
t	wall thickness, m
T	temperature, °C
V	velocity, m/s; volume, m^3
$\dot{W}_p$	pump power, kJ/s
$\dot{Z}$	capital cost rate, US$/s
ϕ	exergy efficiency of pipe segment
x	insulation thickness, m
η	heating system efficiency
σ	Stefan-Boltzmann constant, W/m^{2}K^4
ε	surface emissivity
θ	dimensionless temperature $(T-T_{int})/(T_i-T_o)$

Subscript

1	pipe inside diameter
2	pipe outside diameter
cr	critic
el	electricity
en	energy
f	fluid
i	inside
ins	insulation
int	initial condition
max	maximum
min	minimum
o	outside
opt	optimum
p	pump
s	surface
surr	surrounding
t	total

insulation thickness considers the initial cost of the insulation system plus the ongoing value of the energy savings over the expected service lifetime of the insulation. Numerous studies on thermal insulation systems for different applications, including buildings, cold stores, pipelines, ducts, tanks and other equipment, have been published in the literature. The majority of the studies that are related to optimizing the thermal insulation thickness focus on flat surfaces, such as building walls, rather than cylindrical geometries. The economic insulation thickness for building walls depends on various factors, such as building type, function, wall orientation, construction materials, climatic conditions, insulation properties and cost, energy type and cost and efficiency of the heating or air-conditioning system [4–9]. In these studies, the effect of mass and insulation location on heating and cooling loads were analyzed in buildings with massive exterior envelope components for various wall configurations. Considering the heating and/or cooling loads of buildings, optimization studies on the insulation thickness for the external walls of buildings have been conducted for different climate zones in various countries including Qatar [6], China [9], Maldives [10], Turkey [2,3,10–13], Saudi Arabia [14–16], Malaysia [17], Palestine [18] and Tunisia [19,20]. A suitable amount of thermal insulation in the building envelope can result in a considerable reduction in the heating and cooling energy demands of a building and its associated CO_2 and SO_2 emissions into the atmosphere. In addition to insulation, several investigations have focused on optimizing the building shape to minimize the energy demand and cost [21–23]. All of these studies help to reduce building energy use (annual energy requirements for heating and cooling) and the size of the air-conditioning and heating systems in buildings and to achieve desirable indoor thermal comfort for occupants.

Compared to studies on flat surfaces, there are few studies related to insulation systems on pipes and ducts despite the extensive applications in widely diversified fields. Furthermore, deriving the heat transfer and energy cost equations and taking the derivative of the objective function for cylindrical geometries such as pipes may be considered slightly more difficult. In most studies, optimum insulation thickness computations were performed based mainly on the convective and radiative heat loss from a pipe or duct and other parameters, such as the costs of the insulation material and energy, the heating system efficiency, the lifetime and the current inflation and discount rates. The heat loss from a pipe (or the energy requirement to heat the fluid in a pipe) is the main input required to analyze the optimum insulation thickness. Heat loss occurs by conduction, convection and thermal radiation. Most of the studies considered only convective [24–31] or radiative heat transfer modes [32] while several studies considered both modes [33–36]. Moreover, Kecebas et al. [37] and Basogul and Kecebas [38] used the degree-days (*DD*) for estimating the heating energy requirement.

In this paper, studies related to determining the optimum thermal insulation thickness for pipes and ducts were reviewed first. The optimization procedure was introduced, and the operating conditions and parameters used in these studies were listed. Then, studies determining the optimum insulation thickness on pipes and ducts with various geometries on reducing convective and radiative heat transfer were investigated and presented. After examining the economic analysis methods used to obtain an economic insulation thickness, a simple and practical application for determining the optimum insulation thickness for pipes was conducted.

2. Determining the economic insulation thickness

2.1. Optimization procedure

To minimize the energy and insulation costs in addition to reducing the heat loss to the surroundings, the thickness of the

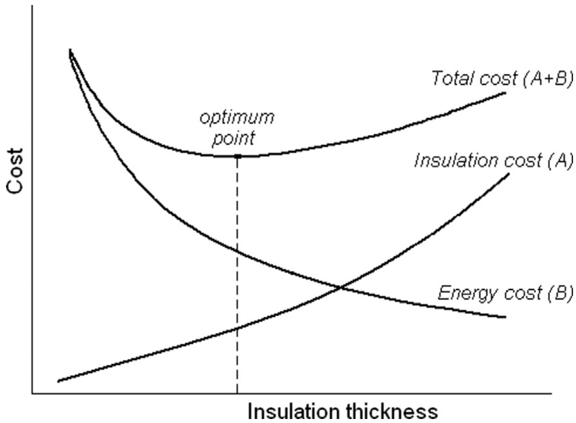

Fig. 1. Optimum thermal insulation thickness.

insulation material needs to be optimized. The motivation for including pipe insulation is often to minimize the total cost, which includes the cost of the insulation, its installation and maintenance as well as the cost of the energy loss by convective or radiative heat transfer. Heat transfer principles and cost information are used in defining the overall cost function to be minimized [33]. The economic insulation thickness for a pipe is a function of a large number of parameters, such as pipe size, cost, conductivities of the pipe and the insulation material, operating and ambient temperatures, heat transfer coefficients at the inside and outside of the pipe, economic parameters and annual operation hours [25,29,38,39]. The optimum economic thickness is generally accepted as the value that provides the minimum total life cycle cost, as illustrated in Fig. 1. As the thermal insulation thickness on a pipe increases, the heating and cooling transmission loads and their costs (i.e., energy costs) decrease. The transmission loads are used as the input data for an economic model to determine the variation in the cost of the insulation plus the present value of the energy consumption, which is considered lost energy, over the lifetime of the system. Conversely, as shown in Fig. 1, the insulation cost increases with the amount of material used. In contrast to flat surfaces, the variation in the amount of insulation used on a pipe versus the insulation thickness is not linear, but rather, the increase is parabolic.

2.2. Optimum thermal insulation thickness for pipes and ducts

Economic evaluations regarding thermal insulation in district heating pipelines were investigated by Kecebas et al. [37] using the heating degree-days (*HDD*) indicator in Afyonkarahisar/Turkey. In the calculations, coal, natural gas, fuel-oil and geothermal energy were used as the energy sources, and rock wool was used as the insulation material for nominal pipe sizes (ranging from 50 mm to 250 mm). Eqs. (1) and (2) were used for the convective heat transfer coefficients inside and outside the insulated pipe, respectively. The results showed that the highest values of the optimum insulation thickness and the energy savings were reached for a nominal pipe size of 250 mm with an insulation thickness of 0.228 m. Similarly, the environmental evaluations of the thermal insulation in the district heating pipeline were investigated by Basogul and Kecebas [38]. Using thermal insulation on pipelines helps to reduce the heating or cooling energy requirements of transmitted fluid and its associated greenhouse gas emissions into the atmosphere. The reductions in the emissions of CO_2, CO and SO_2 with different insulation thickness for nominal pipe sizes ranging from 50 mm to 200 mm were examined under the same conditions as those used in Kecebas et al. [37]. It was found that when the optimum thermal insulation was applied to a heating pipeline, the emission of CO_2 was decreased by 21%.

$$h_i = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}(k_f/D_1) \tag{1}$$

$$h_o = 11.58(1/D_{ins})^{0.2}\left(\frac{2}{(T_{s,o}+T_o)}\right)^{0.181}(T_{s,o}-T_o)^{0.266}(1+2.86V)^{0.5} \tag{2}$$

where D_1 and D_{ins} are the inside pipe and insulation diameters, k_f is the thermal conductivity of the fluid, $T_{s,o}$ is the mean outer surface temperature of the insulation, T_o is the outside air temperature and V is the outside air velocity, which was assumed to be 0.2 m/s.

Ozdemir and Parmaksizoglu [24] focused on determining the optimum insulation thickness for pipes used in low temperature heating systems in buildings. The variation in the annual net savings with the insulation thickness was investigated using the life cycle cost analysis, but the effects of the pipe diameter, the operation period, the operation temperature of water and the properties of the insulation material were not considered in that study. The convective heat transfer coefficient inside the pipe was assumed to be constant and equal to 4000 W/m^{2}K. As opposed to the study by Kecebas et al. [37], the following equations were used for the convective heat transfer coefficient outside the insulated pipe for laminar and turbulent flow, respectively.

$$h_o = 1.25\left(\frac{\Delta T}{2r_{ins}}\right)^{0.25} \tag{3}$$

$$h_o = 8.9\frac{V^{0.9}}{(2r_{ins})^{0.1}} \tag{4}$$

where ΔT is the temperature difference between the outer surface of insulation ($T_{ins,s}$) and the outside air (T_o) in °C, r_{ins} is the insulated pipe radius in m and V is the outside air velocity in m/s.

In the study by Li and Chow [25], the thermal insulation and heating cable systems were used together to protect the water pipes from freezing in cold regions such as Beijing/China. In the calculations, the variation in the optimum insulation thickness was determined depending on the average outside air temperature, the outer pipe diameter, the lifetime, the thermal conductivity of the insulation material and the insulation cost. An empirical formula given by Eq. (5) was used to estimate the convective heat transfer coefficient as a function of the wind speed of the environment. The average outdoor air temperature and the mean wind speed were assumed to be −10 °C and 4.8 m/s, respectively. It was found that increasing the lifetime of the system and decreasing the outside air temperature result in a thicker optimum insulation thickness. Furthermore, the optimum insulation thickness was inversely proportional to the thermal conductivity cost of the insulation.

$$h_o = 11.73 + V^{1/2} \tag{5}$$

For cold applications, the CO_2 emissions and the energy savings potential obtained by using correct pipe insulation in space heating and domestic hot water pipelines were investigated by Chmielarski [40]. In the study, the ambient air temperature, the chilled water supply temperature and the air conditioning period were assumed to be 26 °C, 7 °C and 6 months, respectively. Assuming typical European basic conditions, the optimal insulation thickness varied from 20 mm to 30 mm. Information regarding the studies is summarized in Table 1.

The outside heat transfer coefficient equations for horizontal and vertical steel pipes as a function of the surface temperature of the steel pipe (T_s) and the ambient air temperature (T_o) are given, respectively, as follows [30]:

$$h_o = 3.2 + 0.05(T_s - T_o) \tag{6}$$

Table 1
Summary information about the studies related to the optimization.

Study	Parameters	Economic parameters	Operating conditions	Notes
Kecebas et al. [37]	D_2=50–250 mm, k_{pipe}=16.2 W/mK, k_{ins}=0.040 W/mK (rock wool), C_{ins}=95 US$/m^3	P_1-P_2 method was used. LT=10 years, interest and inflation rates are 4% and 5% respectively.	T_o=15 °C, HDD=2828, energy sources are: coal, natural gas, fuel-oil and geothermal energy.	
Basogul and Kecebas [38]	D_2=50–200 mm, k_{pipe}=54 W/mK, k_{ins}=0.040 W/mK (rock wool), C_{ins}=95 US$/m^3	P_1-P_2 method was used. LT=10 years, interest and inflation rates are 4% and 5% respectively.	HDD=2828, energy sources are: coal, natural gas, fuel-oil, LPG and geothermal.	Environmental impacts were also investigated.
Ozd emir and Parmaksizoglu [24]	D_2=108 mm, k_{pipe}=40 W/mK, k_{ins}=0.04 W/mK (fiberglass)	LT=15 years, interest rate is 5%.	T_f=80 °C, T_o=20 °C, h_i=4000 W/m^{2}K, annual operation time is 5040 h.	Fiberglass and natural gas were assumed as insulation and energy source. The heating system efficiency was assumed to be 0.9.
Li and Chow [25]	D_2=20–200 mm, k_{ins}=0.024 W/mK	LT=5–10 years, interest and inflation rates are 4% and 5% respectively.	T_o=−10 °C, $T_{s,o}$=−26 °C, h_o=13.9 W/m^{2}K (for V=4.8 m/s), 3000 h.	The total cost includes the heating cable cost together with the operating energy and insulation costs.
Kecebas [42]	D_2=50 mm, k_{pipe}=54 W/mK (stainless steel), k_{ins}=0.040 W/mK (rock wool), C_{ins}=95 US $/m^3	LT=10 years, interest and inflation rates are 4% and 5% respectively.	T_f=90 °C	Energy sources are natural gas, coal and fuel-oil. The optimization method is based on exergo-economic analysis.
Chmielarski [40]	D_2=40–115 mm	Energy cost increase is 5%, interest rate is 4%.	T_o=26 °C, T_f=7 °C, operating period 6 months, energy efficiency ratio 2.5.	
Ozturk et al. [36]	$100 < D_2 < 600$ mm, $0 < x_{ins} < 160$ mm, k_{pipe}=54 W/mK, k_{ins}=0.045 W/mK	Interest rate is 6%, LT=20 years.	Fluid is water, T_f=120 °C, T_o=10 °C, energy source is natural gas, annual operation time is 3200 h.	
Soponpongpipat et al. [26]	D_2=0.5 m, k_{duct}=60.5 W/mK, k_{ins}=0.035 W/mK (rubber), k_{ins}=0.045 W/mK (fiberglass)	P_1-P_2 method was used, interest rate is 14.25%, inflation is 2.1%, LT=10 years.	T_f–T_o=13 °C, fluid is air, average COP of air conditioning system is 2.81, total operation time 23040 h.	Rubber and fiberglass were assumed as main and auxiliary insulation materials, respectively.
Zaki and Al-Turki [29]	D_2=10–27.3 cm, insulation materials: rock wool and calcium silicate.	Interest rate was not given.	8760 h, h_o=10 W/m^{2}K, T_f=170–400 °C.	Superheated systems were studied, such as steam, furfural, crude oil and 300-distillate oil.
Soylemez [43]	C_{duct}=10 $/kg, C_f=0.075 $/kWh (electricity)	P_1-P_2 method was used, energy price and interest rate are 0.1, LT=20 years.	Annual operating hours is 1600 h.	A present sample problem was used for comparison.
Kalyon and Sahin [27]	k_{ins}=0.036 W/mK, k_{pipe}=30 W/mK, r_{pipe}=0.0127 m, t_{pipe}=0.002 m	Economic analysis was not used.	h_o=10 W/m^{2}K, T_o=0 °C, T_f=77 °C.	Only convective heat transfer was considered.
Sahin [32]	k_{ins}=0.036 W/mK, k_{pipe}=30 W/mK, r_{pipe}=0.0127 m, t_{pipe}=0.002 m	Economic analysis was not used.	T_o=−273 °C, T_f=77 °C, h_i=122 W/m^{2}K, ε=0.80.	Only radiative heat transfer was considered.
Sahin and Kalyon [34]	k_{ins}=0.01–0.05 W/mK, k_{pipe}=30 W/mK, r_{pipe}=0.0127 m	Economic analysis was not used.	h_o=10 W/m^{2}K, h_i=102.6 W/m^{2}K, T_o=0 °C, T_f=77 °C.	Convective and radiative heat transfer were considered.
Bahadori and Vuthaluru [44]	D_2=0.02–0.5 m, k_{ins}=0.02–0.09 W/mK	Economic analysis was not used.	T_s=100–700 °C, T_o=20 °C	The average absolute deviation is approximately 2.12% depending on surface temperature and diameter.
Bahadori and Vuthaluru [45]	$100 < D_2 < 2400$ mm	Economic analysis was not used.	T_s–T_o < 250 °C	The average absolute deviation is approximately 3.25%.
Sahin et al. [46]	D_2=0.1 m, k_{pipe}=0.144 W/mK	The costs of irreversibilities due to the heat transfer and pressure drop were computed.	T_s=350 K, T_o=300 K, L=100 m, k_f=2.25 W/mK (engine oil)	

$$h_o = 3.4 + 0.09(T_s - T_o) \tag{7}$$

Ozturk et al. [36] compared four optimization methods for the design of hot water piping systems. The aim of the first method is to optimize both the pipe diameter and the insulation thickness. First, the pipe diameter is optimized on the basis of the pipe investment cost and the operation cost due to the friction of the fluid in the pipe. Then, the insulation thickness on a pipe is optimized by considering energy and insulation costs. The operation cost is related to the power required to pump the fluid along the piping. The insulation cost is a function of the thickness of the insulation that is applied on the pipe. The energy cost is a concept that is related to the energy source type and the heat loss per length of pipe. In Ozturk et al. [36], natural gas was used as an energy source. The objective functions to optimize the pipe diameter and the insulation thickness are given, respectively, as follows:

$$\dot{C}_{pipe} = \dot{Z}_{pipe} + (C_{el}\dot{W}_p) \tag{8}$$

$$\dot{C}_{ins} = \dot{Z}_{ins} + (C_h Q) \tag{9}$$

where $\dot{C}$ is the cost rate in US\$/s, $\dot{Z}$ is the capital cost rate in US\$/s, C_{el} and C_h are the specific cost for electricity and the unit energy cost of hot water in US\$/kJ, respectively, $\dot{W}_p$is the pump power in kJ/s and Q is the rate of heat loss from the pipe in kJ/s. The capital cost rates for the pipe and the insulation can be expressed by Eqs. (10) and (11) as follows:

$$\dot{Z}_{pipe} = (CRF + \varsigma)C_{pipe} \tag{10}$$

$$\dot{Z}_{ins} = (CRF + \varsigma)C_{ins} \tag{11}$$

where *CRF* is the capital recovery factor, which is dependent on the annual interest rate and the lifetime (*LT*), ς is a coefficient that accounts for part of the fixed operation and the maintenance cost, which was assumed to be 0.01, and C_{pipe} and C_{ins} are the pipe and insulation costs, respectively.

The second method considers not only the insulation but also the pipe system, and it is based on the minimization of the total cost. The total cost includes the pipe and insulation investment costs and the operating costs due to pressure and heat losses, which is given by

$$\dot{C}_t = (\dot{Z}_{pipe} + \dot{Z}_{ins})(C_{el}\dot{W}_p + C_h Q) \tag{12}$$

The third and fourth methods mentioned in Ozturk et al. [36] involve an exergy analysis for the water stream in a pipe. The third method is focused on the minimization of entropy generation or the maximization of exergy efficiency without cost effects. The objective of the method is to increase the coefficient of thermal performance (*CTP*), and which is given as follows:

$$CTP = \frac{\phi}{(1-\phi)} \tag{13}$$

where ϕ is defined as the exergy efficiency of the pipe segment, which is the ratio of the exergy flow at the pipe outlet to the exergy flow at the pipe inlet plus the work supplied for hot water transportation. The pipe diameter and the insulation thickness are determined considering the maximum *CTP*.

The fourth and most recent method mentioned in Ozturk et al. [36] considers the costs of exergy destruction due to the friction loss and the exergy loss from heat loss in addition to the pipe and insulation investment costs. The objective function used to determine the economic sizes of the pipe and the insulation is given by the following:

$$\dot{C}_t = (\dot{Z}_{pipe} + \dot{Z}_{ins})(C_{el}\dot{E}_{dest} + C_e\dot{E}_{loss}) \tag{14}$$

where C_{el} and C_e are the specific exergy costs for electricity and hot water, respectively, $\dot{E}_{dest}$ and $\dot{E}_{loss}$ are the exergy destruction rate due to friction in the pipe and the exergy loss rate due to heat loss from the pipe to the environment, respectively. The details of the equations can be found in Ozturk et al. [36] and Karabay [41]. Using these equations, Ozturk et al. [36] compared the results of these methods when using the same criteria in a case of a pipe system. Several parameters used in that study and their numerical values are given in Table 1. As a conclusion, in the study, the fourth method was recommended to be used in the design studies because the exergy and cost parameters were considered. Similarly, in the study of Kecebas [42], the exergoeconomic approach was used for optimizing the pipe insulation thickness. The exergetic costs of insulation and fuels (coal, natural gas and fuel–oil) were calculated based on life-cycle cost analysis. The parameters considering in the analysis were excess air coefficient, the inlet temperature of fuel, the temperature of combustion chamber and the temperature of stack gases. It was stated in the study that the optimum insulation thickness for the exergoeconomic optimization was higher than that of energoeconomic optimization.

Soponpongpipat et al. [26] studied the economic thickness of a double-layer insulation composed of the rubber (k=0.035 W/mK) and fiber glass (k=0.045 W/mK) for an air conditioning duct. The effects of the convective heat transfer coefficients of the inside and outside of a circular galvanized steel duct on the optimum thickness of these insulators were investigated when the heat transfer coefficients varied from 6 W/m^{2}K to 22 W/m^{2}K. It was found that the variation in the inside and outside heat transfer coefficients did not affect the optimum thickness, but the net savings increased with increasing the heat transfer coefficients.

Zaki and Al-Turki [29] presented a model for calculating the optimum thickness of an insulation system in which the composite materials have different properties and costs for pipes (diameter of 10 cm –27.3 cm) with a flow of superheated steam. Mineral rock wool and calcium silicate were assumed to be the insulation materials.

In addition to optimization of insulation thickness, Soylemez [43] studied the optimum duct sizing for heating, ventilation and air conditioning (HVAC) systems with round and rectangular shapes. It was stated that the airflow velocity in an HVAC channel is directly related to the operational cost of the HVAC system along with the initial system cost. The derivative of the total life cycle cost, which is the sum of the initial cost and the operating (energy) cost, was taken with respect to the pressure drop per unit channel length. The optimum duct diameter was determined to obtain the minimum pressure drop. In the optimization, the P_1–P_2 economic method was used and the technical lifetime of the duct was assumed to be 20 years. Sahin et al. [46] numerically investigated the effect of fouling thickness, which leads to decreased heat transfer and increased frictional pressure drop, on the operational cost related to irreversibilities in pipe flow. The numerical model developed was based on entropy generation due to heat transfer and viscous friction.

All of the studies mentioned above are given in Table 1, which describes the parameters, assumptions and operating conditions of the systems.

2.3. Studies not including an economic analysis

Heat transfer modes depend mainly on the surface temperatures and the geometry of the system being considered. Pipes or ducts surrounded by gases may transfer heat via conduction, convection, and/or thermal radiation. For air, conduction is usually dominated by other modes and thus, may be neglected [47]. Kalyon and Sahin [27] studied the optimum insulation thickness of a circular pipe subjected to convective heat transfer. The objective is to determine the insulation thickness for a limited amount of insulation material, which minimizes the heat loss.

For this reason, the mentioned study did not use any economic methods. They found that a uniform insulation layer on the surface of a pipe provides optimum results for the case of a convective heat transfer mode. Sahin [32] numerically optimized the variation in the thermal insulation thickness of a pipe for space applications to minimize the radiative heat loss to the ambient. The thickness of the insulation was assumed to be linearly varying along the pipe because this scenario is easy to implement in practice. The study aimed to determine the suitable slope of the insulation thickness along the pipe, which maximizes the fluid outlet temperature (thus minimizes the heat loss). It was found that a linearly decreasing insulation thickness with a minimum slope along the pipe provides the best insulation under the radiation heat transfer condition.

As reported in Sahin and Kalyon [34], the weight and volume of the insulation material are required to be minimal in some cases, such as in aerospace applications. For this reason, they studied the insulation thickness variation over a pipe transporting a high temperature fluid to maintain a uniform outer surface temperature. The heat transfer on the outer surface of the insulation was considered to be a combination of convection and radiation. In the study, the variation in the insulation thickness over the pipe with a dimensionless axial distance was calculated. The analytical solution showed that the insulation thickness variation providing the uniform surface temperature is independent of the surrounding thermal conditions (i.e., the convection and radiation heat transfer coefficients) provided that these conditions are uniform over the surface. Moreover, the insulation thickness variation along a pipe was determined in an exponential form (decreasing in an exponential function); however, it was very close to a linear variation.

Bahadori and Vuthaluru [44] formulated a simple correlation to estimate the economic thickness of the thermal insulation for pipes. The proposed correlation depends on the steel pipe diameter and the thermal conductivity of the insulation and covers a pipeline diameter and a surface temperature of up to 0.5 m and 700 °C, respectively. Similarly, Bahadori and Vuthaluru [45] developed a simple method to estimate the thermal insulation thickness required to arrive at a desired heat flow or surface temperature for flat surfaces, ducts and pipes. The proposed method considers temperature differences between the ambient and outside temperatures up to 250 °C, the temperature drop through the insulation up to 1000 °C and the outside diameter of a pipe up to 2400 mm. It was noted in the study that the results of the model are in good agreement with the reported data for a wide range of conditions, and the average absolute deviation was found to be 3.25% for a temperature drop between 25 °C and 1000 °C through the insulation.

Alawadhi [31] investigated the use of a phase change material (PCM) as thermal insulation for a pipe. The effectiveness of the PCM insulation was evaluated by comparing its thermal performance with that of insulation without a phase change. N-octadecane ($C_{18}H_{38}$) paraffin was selected as the PCM. Both time-dependent and time-independent boundary conditions were examined. At the outer surface, the radiation effect was not considered and the free convection boundary condition ($\theta_o=0$, Bi$=0.5$) was assumed. The results indicated that for the time-independent case, the PCM insulation is effective in reducing the heat loss from the pipe if the Rayleigh number is low (Ra$=5\times10^6$); but for Ra$=1\times10^7$, the PCM insulation shows a weak thermal performance. For the time-dependent case, the heat loss is effectively reduced with the PCM insulation for the examined range of oscillating periods.

2.4. Energy, insulation and total cost equations

The total thermal resistance for the circular pipe illustrated in Fig. 2 can be written as follows:

$$\Sigma R=\left(\frac{1}{2\pi r_1 L h_i}+\frac{In(r_2/r_1)}{2\pi k_{pipe}L}+\frac{In(r_{ins}/r_2)}{2\pi k_{ins}L}+\frac{1}{2\pi r_{ins}L h_o}\right) \tag{15}$$

where r_1 and r_2 are the inner and outer radii of the pipe, respectively, h_i and h_o are the convective heat transfer coefficients of the inner and outer surfaces of the pipe, respectively, L is the length of the pipe, r_{ins} is the radius of the insulated pipe, k_{pipe} and k_{ins} are the thermal conductivities of the pipe and insulation materials, respectively. The rate of the heat loss from the pipe per unit length can be calculated as follows [27,32,33,35,37,48]:

$$\frac{Q}{L}=\frac{2\pi\Delta T}{(1/r_1h_i)+(In(r_2/r_1)/k_{pipe})+(In(r_{ins}/r_2)/k_{ins})+(1/r_{ins}h_o)}=\frac{(T_f-T_o)}{\Sigma R} \tag{16}$$

where T_f and T_o are the temperatures of the fluid in the pipe and the outside air.

The energy cost due to the heat loss rate is given as follows:

$$C_{en,t}=\frac{C_{en}2\pi 3600\Delta TN}{((1/r_1h_i)+(In(r_2/r_1)/k_{pipe})+(In(r_{ins}/r_2)/k_{ins})+(1/r_{ins}h_o))Hu\,\eta} \tag{17}$$

where C_{en} is the unit cost of the energy source in US$/kg, US$/m^3 or US$/kWh, depending on the energy source type, N is the annual operating time in hours, Hu is the heating value of the fuel in kJ/kg or kJ/m^3, depending on the energy source type and η is the heating system efficiency, which was generally assumed to be 0.90 or 0.93 for natural gas usage [2,11,24,36–38,49,50], 0.65, 0.70 or 0.77 for coal [2,11–13,37,38,51], 0.80 for fuel oil [2,11,30,37,38,50] and 0.99 for electricity [2,11].

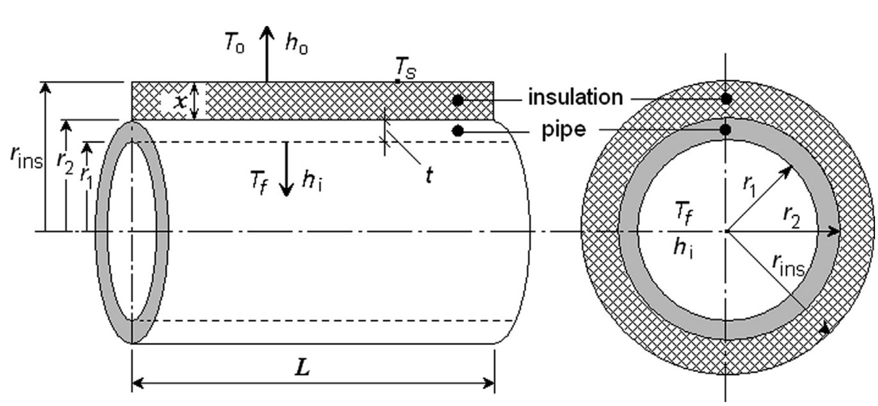

Fig. 2. An insulated pipe.

The amount of used insulation per unit length of pipe and its cost are, respectively, given as follows:

$$\frac{V_{ins}}{L} = \pi(r_{ins}^2 - r_2^2) \tag{18}$$

$$C_{ins,t} = C_{ins}(V_{ins}/L) = C_{ins}\pi(r_{ins}^2 - r_2^2) \tag{19}$$

where V_{ins} is the volume of insulation material used in m^3, C_{ins} is the unit cost of the insulation in US\$/m^3 and $C_{ins,t}$ is the cost of insulation per unit length of pipe in US\$/m.

After calculating the energy and insulation costs, the present worth of the total life cycle cost associated with the pipe insulation thickness can be expressed as follows:

$$C_t = \frac{C_{en}2\pi 3600\,\Delta T\,N\,PWF}{((1/r_1h_i)+(In(r_2/r_1)/k_{pipe})+(In(r_{ins}/r_2)/k_{ins})+(1/r_{ins}h_o))Hu\,\eta} + C_{ins}\pi(r_{ins}^2 - r_2^2) \tag{20}$$

where *PWF* is the present worth factor, the formulation of which is given in the Economic Analysis section. The optimum insulation thickness is obtained by minimizing the total cost. The minimum point can be determined by setting the first derivative of the total cost function to zero with respect to the radius of the insulated pipe. Consequently, the partial derivative of C_t with respect to r_{ins} can be computed ($\delta C_t/\delta r_{ins}$). Equating the first derivative function to zero yields the value of $r_{ins,opt}$, which is determined as follows:

$$0 = (h_o r_{ins,opt} - k_{ins})^{1/2}/(r_{ins,opt})^{3/2}((3600C_{en}(\Delta TN)PWF)/(Hu\eta k_{ins}h_oC_{ins}))^{1/2} - ((1/r_1h_i)+(In(r_2/r_1)/k_{pipe})) - ((In(r_{ins,opt}/r_2)/k_{ins})+(1/r_{ins,opt}h_o)) \tag{21}$$

Hence, the optimum insulation thickness for a pipe can be written as follows:

$$x_{opt} = r_{ins,opt} - r_2 \tag{22}$$

3. Studies on heat transfer characteristics in ducts and pipes

Several investigations have focused on the heat transfer characteristics in various geometries (circular, triangular, rectangular, oval, polygonal ducts and double pipes) rather than optimizing the insulation thickness by considering the economic parameters [28,33,35,48,52–55]. Chen and Yang [54] investigated the heat transfer rate on the insulation layer of a double circular pipe and the temperature distribution in the double pipe. Wong et al. [55] analyzed the two dimensional steady state heat transfer characteristics of an insulated triangular duct by using the two models, which are the plane wedge thermal resistance and plate thermal resistance models. The critical and neutral thicknesses as well as heat transfer characteristics for an insulated pipe have been interesting subjects to researchers. The critical and neutral heat transfer characteristics were investigated by Chou [52] and Wong et al. [53] for polygonal (and circular) ducts and by Wong et al. [28] for oval ducts with different long–short axes ratios. The critical and neutral radii (r_{cr} and r_e) are defined, respectively, as the outer radius of the insulation for which the heat transfer rate from the circular duct is a maximum, Q_{max} (or the overall thermal resistance is a minimum, ΣR_{min}), and that for which the heat transfer rate from the circular duct is the same as that of a non-insulated duct ($Q_e = Q_o$), as illustrated in Fig. 3. Critical heat transfer occurs when the outer radius of an insulated circular duct is less than its critical insulation radius. This phenomenon, according to Wong et al. [28], is important and also occurs in the case of a small-sized insulated duct, especially in conditions of low ambient air/gas convective coefficients (h_o). The study showed that an insulated oval duct is similar to an insulated circular duct except its critical and neutral thicknesses are smaller in the larger oval axes ratio, i.e., less

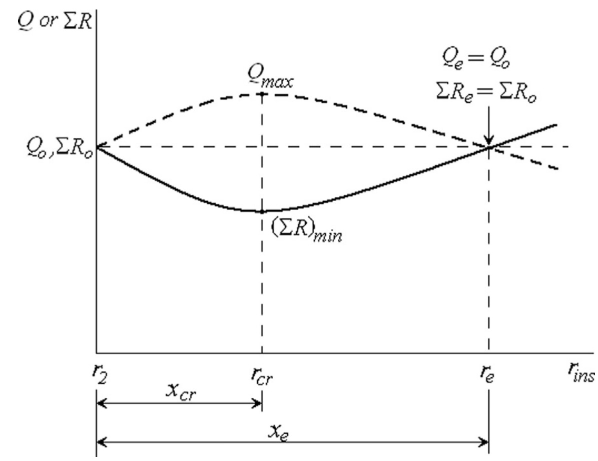

Fig. 3. The critical and neutral heat transfer phenomena [48,3]. Q_o, heat transfer rate from bare pipe; r_{cr}, critical radius; r_e, insulated pipe radius, where Q is the same as that of a non-insulated pipe; r_{ins}, insulated pipe radius; and ΣR, overall thermal resistance.

insulation material would be required if the long–short axes ratio of the oval duct increases (i.e., for a flat–oval duct). Sahin and Kalyon [33] specified that the critical radius analysis of a circular pipe for convection-type heat transfer is well known, and the analysis was extended to radiation heat transfer applications. In that study, a circular pipe in which the outer surface is subject to both convection and radiation with the surroundings was handled, and a possible critical radius for various parametric conditions was studied. They found that the critical radius of the insulation can be calculated for a circular pipe subjected to radiative and convective heat transfer environments provided that certain constraints are satisfied and the critical radius of the insulation increases when either the convection or radiation decreases. A comprehensive study on a critical radius analysis of pipes and ducts was carried out by Aziz [35] in which both cylindrical and spherical geometries were studied. In addition, cylindrical, spherical, rectangular, equilateral polygonal and eccentric circular insulation geometries were also examined. The heat transfer characteristics for insulated and non-insulated spherical containers were investigated by Wong et al. [56]. By both considering and neglecting the effect of radiative heat transfer on a spherical geometry were studied. They found that the effect of radiative heat transfer cannot be ignored in conditions of low outside convective heat transfer coefficients (such as ambient air) and high surface emissivities. x_{ins}/r_2 (insulation thickness divided by outside radius of sphere) $=0.2$ was suggested for optimum insulation thickness in practical applications because of that the difference of heat radiation between the insulated and non-insulated cases was very small (within 3%) for between $x_{ins}/r_2 = 0.2$ and $x_{ins}/r_2 = 1$. Information about these studies and the particular data used are summarized in Table 2.

The performance of an insulation material is mainly determined by its thermal conductivity, which is dependent on the density, porosity, moisture content, and mean temperature difference of the material. Thermal insulation materials in various applications are exposed to significant and continuous temperature variations due to a varying ambient air temperature, operating temperature and solar radiation. Therefore, the thermal conductivity of the insulation material can vary due to changes in both moisture content and temperature. Condensation in the insulation occurs if the water vapor concentration reaches saturation. Condensation in fibrous insulation is a serious problem in many applications such as building insulation, refrigerated space envelopes and pipe/duct insulation, because it can result in a drastic reduction in the thermal insulation of the system. The heat

Table 2
Brief information about the studies analyzing the critical radius of insulation.

Study	Parameters	Studied geometries	Operating conditions	Notes
Chou and Wong [48]		Regular polygonal (triangular, square, pentagonal etc.) pipes	Inner convection coefficient (h_i) and pipe conductivity (k_{pipe}) were neglected.	The critical and neutral thicknesses were studied.
Wong et al. [53]	$k_{ins}=0.04$ W/mK	Regular polygonal (triangular, square, pentagonal) and circular pipes	Low inner convection coefficients ($h_i=5$–83.5 W/m^{2}K) and low pipe conductivities ($k_{pipe}=1$–10 W/mK), $h_o=5$ W/m^{2}K	The critical and neutral thicknesses were studied.
Wong et al. [28]	$k_{duct}=77$ W/mK, $r_2=2.13$–3.96 mm, $k_{ins}=0.035$ W/mK	Oval duct, long–short axes ratios vary between 1 and 6.	$T_i=100$ °C, $T_o=0$ °C, $h_i=10^5$ W/m^{2}K, $h_o=8.3$ W/m^{2}K	Optimization was not performed. Heat transfer characteristic for insulated oval ducts was investigated for different axes ratios.
Sahin and Kalyon [33]		Circular ducts	$T_o/T_f=0$–0.3, $(h_o r/k)=0.2$–1, $(\sigma\varepsilon r T_f^3/k)=0.1$–$0.2$	Radiative heat transfer in addition to convective for insulated circular pipe was considered and critical radius in terms of heat transfer rate was studied.
Aziz [35]	Generally, $k_{ins}=0.18$ W/mK (polystyrene), $k_{ins}=0.16$ W/mK (rubber), $r_{pipe}=0.25$–4 cm.	Cylindrical, equilateral polygonal, rectangular and eccentric circular insulation shapes	$T_s=80$ °C, $h_o=3.0$–8.52 W/m^{2}K and $h_o=1.109(T_s-T_o)/r_{ins}^{1/4}$ for natural convection, $h_o=22.7$ W/m^{2}K and $h_o=3.246/r_{ins}^{0.382}$ for forced convection, $\varepsilon=0$–0.9.	Both convection (natural and forced) and radiative heat transfer modes were considered.
Wong et al. [56]	$k_{ins}=0.035$ W/mK, $k_{pipe}=77$ W/mK (carbon steel), $r_1=190$ mm, $r_2=200$ mm	Spherical geometries	$T_i=100$ °C, $T_o=30$ °C, $T_{surr}=30$–32 °C, $h_o=10$ W/m^{2}K for hot conditions; $T_i=-20$ °C, $h_o=8.3$ W/m^{2}K for cold conditions.	Only radiative heat transfer was considered.

and moisture transfer characteristics in fibrous insulation were investigated by Leskovsek and Medved [57], Fan and Wen [58] and Choudhary et al. [59]. Leskovsek and Medved [57] found that a relatively small mass of water in the insulation matrix can result in a significantly increased average heat flux. Moreover, Fan and Wen [58] found that the initial water content, the thickness of the fibrous insulation and the environmental temperature are the three most important factors influencing the heat flux. Choudhary et al. [59] studied the condensation around a cold pipe and the removal of the condensate by a hydrophilic wick fabric with downward orientation by the combined effect of capillary and gravity actions. As a result, the presence of the wick appears to significantly reduce the amount of liquid water in the insulation and to maintain the conductivity of the insulation.

4. Economic analysis

In the literature, several economic analysis methods are used to optimize the thermal insulation thickness for different applications, such as walls, pipes and ducts, and to calculate the payback period. One of the economic methods is the Simple Payback Analysis. This method is based on the time required to repay the initial capital investment with the operating savings attributed to that investment. The main drawback of this simple analysis is that it does not take into account the inflation and the time value of money, which are very important financial considerations [60,61]. An economic assessment is generally performed using a life-cycle cost (LCC) method during each period of time. In this analysis, the amount of energy savings due to the insulation over a lifetime is evaluated in the present value using the present worth factor (*PWF*). The *PWF*, which depends on the inflation rate (i) and the discount rate (d), is calculated as follows [18,25,62]:

$$PWF = \left(\frac{1+i}{d-i}\right)\left[1-\left(\frac{1+i}{1+d}\right)^{LT}\right] \qquad (\text{if } d \neq i) \tag{23}$$

$$PWF = \frac{LT}{1+d} \qquad (\text{if } d = i) \tag{24}$$

The *LT* could be assumed to be 10 years [25,26,37,38], 15 years [24] or 20 years [36].

The P_1–P_2 method presented in Duffie and Beckman [63] is used in several studies [26,37,38]. This method simplifies the economic analysis by concentrating all of the economic parameters into two parameters, P_1 and P_2. P_1 and P_2 are determined from the economic parameters (e.g., interest rate, inflation rate, period of economic analysis). P_1 is the life cycle energy related to the market discount rate d (for the value of money), the inflation rate i (for the energy cost), and the economic analysis period (*or* the technical lifetime of the applied insulation in years *LT* [26,64]). The value of P_1 can be calculated by the following equation:

$$P_1 = \sum_{j=1}^{LT} \frac{(1+i)^{j-1}}{(1+d)^j} = \left| \begin{array}{ll} \frac{1}{(d-i)}\left[1-\left(\frac{1+i}{1+d}\right)^{LT}\right] & i \neq d \\ \frac{LT}{1+i} & i = d \end{array} \right. \tag{25}$$

Typically, for a period of economic analysis of *LT* years, the value of P_1 varies between $LT/2$ and LT. P_2 is the ratio of the life cycle expenditures incurred due to the additional capital investment to the initial investment, which can be calculated by the following equation:

$$P_2 = DP + (1-DP)P_1 + M_s P_1 - \frac{R_v}{(1+d)^{LT}} \tag{26}$$

where *DP* is the ratio of the down payment to the initial investment, M_s is the ratio of first year miscellaneous costs (maintenance, insurance, and other incidental costs) to the initial investment, and

R_v is the ratio of the resale value at the end of the economic period to the initial investment. Typically, the value of P_2 varies between 0.5 and 1.0. If no additional capital is invested other than the initial investment (such as maintenance and operation costs), P_2 can be set to 1.0 [17,37,38,64].

It is noted that because the chosen economic parameters can strongly affect the present worth of the total life cycle cost (and thus, the optimum thermal insulation thickness), they must be accurately determined [47].

5. Investigation of parameters affecting optimum thickness

A practical application example for optimizing the thermal insulation thickness on a pipe is conducted using the P_1–P_2 method, and the factors affecting the economic thickness are investigated in the present study. In the present analysis, a circular pipe subjected to convective heat transfer to the surrounding environment is studied. It is assumed that the carrier pipe has a 100 mm nominal diameter (112 mm outer diameter), is insulated with mineral wool with a thermal conductivity of 0.040 W/mK and operates at 120 °C for 3000 h per year. The useful lifetime of the system is assumed to be 10 years [25,26,37,38], and the inflation and discount rates are 4% and 5%, respectively. The convective heat transfer coefficient at the surface of the insulation is set to a typical value of 10 W/m^{2}K [29,34,47]. The particular parameters used in the calculations are presented in Table 3. As the insulation thickness on a pipe increases, the heat loss per unit length of pipe decreases (and thus, the energy requirements), as expected. The variation in the cost curves (energy, insulation and total) with the thermal insulation thickness and the optimum economic thickness of the insulation can be observed in Fig. 4. With an increasing insulation thickness, the total life cycle energy cost decreases. In contrast to flat surfaces, such as the external walls of a building, the increase is not linear because of the amount (volume) of insulation used on the pipe. The optimum insulation thickness that minimizes the total life cycle cost over a lifetime of 10 years is found to be 0.1136 m.

The variations in the optimum insulation thickness with the convective heat transfer coefficients inside and outside of the pipe are shown in Figs. 5 and 6, respectively. Typical value ranges of the forced convective heat transfer coefficient inside the pipe for air or water flow are also given in Fig. 5. Furthermore, for a pipe surrounded by air, typical convective heat transfer coefficients at the outer surface of the insulation for free and forced modes, which are related to the wind velocity, are also shown on Fig. 6. As shown in the figures, the optimum insulation thickness increases with increasing heat transfer coefficients. Insulation provides the primary thermal resistance against heat loss. Therefore, if the heat transfer coefficient increases, the total thermal resistance of the pipe decreases, and this results in an increased insulation thickness. The variation in the insulation thickness is higher at low heat transfer coefficients. Because the internal heat transfer coefficient is influenced by the fluid properties, the type of working fluid in the pipe (air, water, etc.) is essential. Moreover, because low heat transfer coefficients inside the pipe or the duct are generally encountered for gas flows such as air, the flow velocity of the gas fluid in a pipe or a duct should be determined. In general, the effect of the inner heat transfer coefficient on the optimum insulation thickness is slightly lower than that of the outer heat transfer coefficient, as seen in Figs. 5 and 6.

The working fluid temperature in a pipe is one of the important factors affecting the required insulation thickness. Fig. 7 presents the effect of the fluid temperature on the optimum insulation thickness for different thermal conductivities of insulation materials. In the calculations, the values of the other parameters were taken as given in Table 3. Because an increase in the fluid temperature and in the conductivity of the insulation leads to an increase in the heat loss rate from the pipe, a greater amount of

Table 3
The parameters used in the calculations.

Parameter	Value
Pipe	
Inner radius, r_1	0.05 m
Wall thickness, t	0.006 cm
Outer radius, r_2	0.056 m
Thermal conductivity, k_{pipe}	54 W/mK [36]
Inner heat transfer coefficient, h_i	100 W/m^{2}K [34]
Outer heat transfer coefficient, h_o	10 W/m^{2}K [29,34,47]
Working fluid temperature in pipe, T_f	120 °C
Outside air temperature, T_o	15 °C
Energy source (natural gas)	
Lower heating value, Hu	34526 kJ/m^3
Cost, C_{en}	0.4 US$/m^3
System efficiency, η	0.93
Thermal insulation (mineral wool)	
Thermal conductivity, k_{ins}	0.04 W/mK [37,47]
Cost, C_{ins}	140 US$/m^3
Economic parameters	
Inflation rate, i	0.04
Discount rate, d	0.05
Lifetime, LT	10 [25,26,37,38]
Operating time, N	3000 h

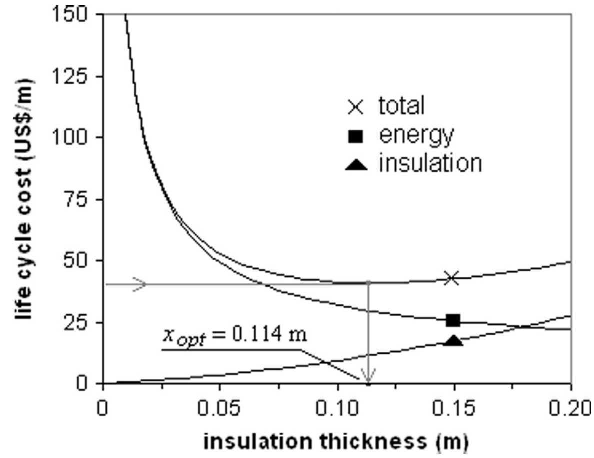

Fig. 4. The variation in life cycle costs with insulation thickness.

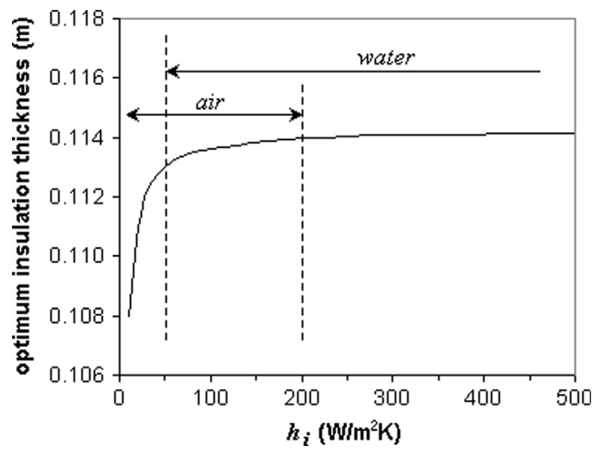

Fig. 5. The optimum insulation thickness vs. the inside heat transfer coefficient.

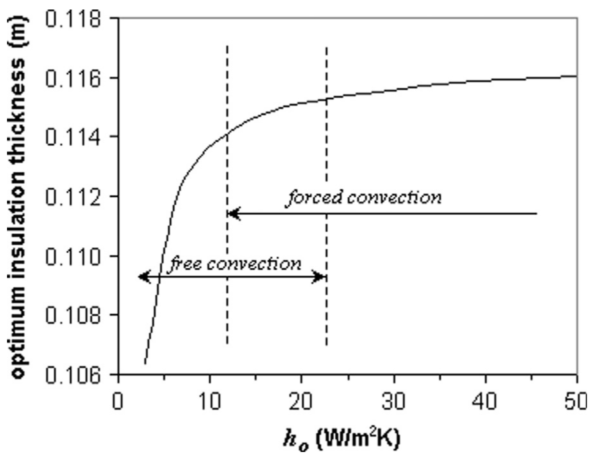

Fig. 6. The optimum insulation thickness vs. the outside heat transfer coefficient.

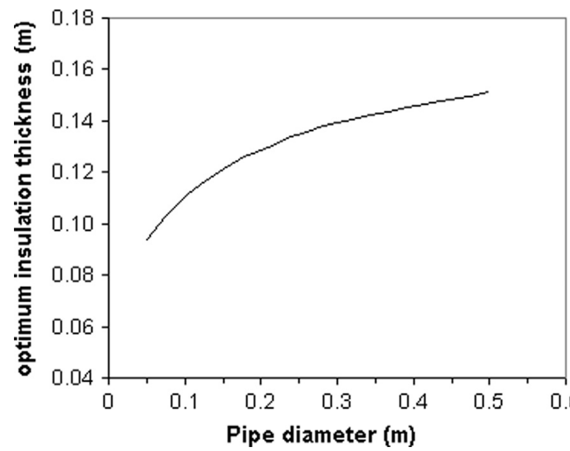

Fig. 9. The optimum insulation thickness vs. pipe diameter (D_2).

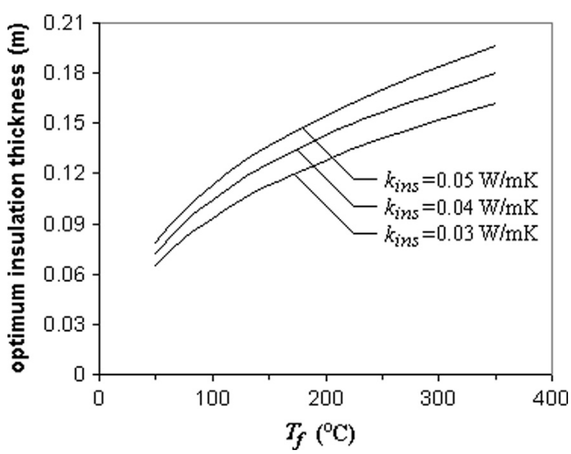

Fig. 7. The optimum insulation thickness vs. the fluid temperature in a pipe.

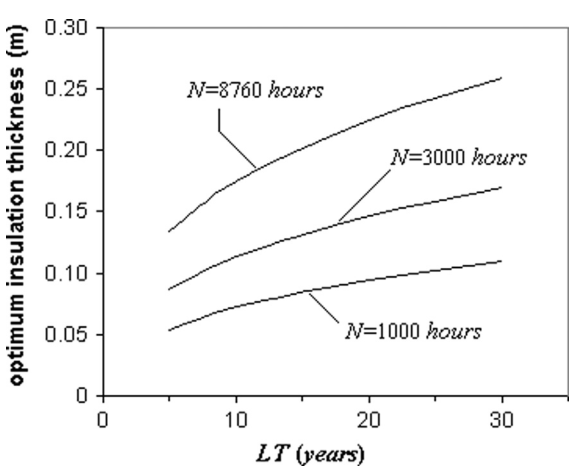

Fig. 8. The optimum insulation thickness vs. lifetime.

insulation is required. Thus, the economic insulation thickness increases with an increase in the fluid temperature and in the thermal conductivity of insulation.

The expected useful lifetime of a system and the operating time in a year are among the important factors in an economic analysis. In the literature, the lifetime was generally assumed to be 10 years; however, various periods ranging from 5 years to 30 years were also used [19,20,24,25,36]. To investigate the influence of the operating time in a year, a wide range from 1000 h to 8760 h, which covers the entire year, is considered. Fig. 8 shows that the optimum insulation thickness increases with an increase in both the lifetime and the annual operating hours. The increases are not linear; therefore, as the lifetime or the operating time increases by three times (from 10 years to 30 years or from 1000 h to 3000 h), the optimum thickness does not increase in the same ratio. Moreover, the annual operating time is slightly more effective than the lifetime on the optimum insulation thickness. The effect of pipe's outer diameter on the optimum insulation thickness is shown in Fig. 9. In the calculations, the values of the parameters were taken as given in Table 3, except for the value of the pipe diameter. Because the heat transfer surface area increases with an increasing pipe diameter, the required insulation thickness increases to limit the heat transfer rate. The increment has a decreasing slope, as shown in Fig. 9. Therefore, as the pipe diameter increases, a relatively thinner insulation thickness is required.

6. Conclusions

In recent years, a significant increase in the number of studies that have been done on the performance of thermal insulation systems applied to flat surfaces, cylindrical surfaces and different geometries has been observed. Reducing the energy consumption in industry is important when considering the limited energy resources and the environmental concerns. This study presented a literature review on the thermal insulation applications in pipes and ducts, and focused particularly on the determination of an optimum insulation thickness considering economic criteria. The insulation materials, the fuel types, the pipe/duct geometries, the fluids types and the operating conditions encountered in the literature were compiled and presented. The effects of thermal insulation and other design and operating parameters on the energy loss from pipes and ducts were investigated. This study could serve as a useful source of information for researchers because it includes a review of the thermal insulation applications particularly in pipe and duct systems, the optimization procedure for determining the economic insulation thickness and a practical application example.

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